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Heat Pump Application for Newsprint Machine

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H.A. Simons (International) Ltd.

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HEAT PUMP APPLICATION FOR NEWSPRINT MACHINE

PREPARED
FOR

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NATIONAL RESEARCH COUNCIL
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By:

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ABSTRACT

A Heat Pump system application is analysed for recovery of thermal energy from the News Machine hood exhaust after the "conventional" heat recovery, with an objective to produce low pressure steam to substitute boiler steam used for Hood Make-up Air and Pocket Ventilation Air Heaters.

This is achieved by a refrigerant cycle generating sub-atmospheric pressure steam, which is then compressed to the desired pressures by the centrifugal compressors.

RÉSUMÉ

On étudie la possibilité d'employer un système à pompe à chaleur pour récupérer, après récupération "classique", l'énergie thermique de la sortie de la hotte d'une presse à journaux, l'objectif étant de produire de la vapeur basse pression qui remplacerait la vapeur de chaudière alimentant les réchauffeurs de l'air d'appoint de la hotte et l'air du ventilateur de poche.

Cela sera réalisé à l'aide d'un circuit de réfrigération qui produira de la vapeur sous pression subatmosphérique qui sera alors amenée à la pression voulue par compression dans des compresseurs centrifuges.

PREFACE

The Study is specifically developed for the conditions of the No. 3 News Machine of the James Maclaren Industries, Inc., Masson, Quebec, and has been reviewed by our MR. R.L. BOEHM, PhD, Paper Production Specialist, and MR. J.D. THOMPSON, P.Eng., Manager HVAC Dept.

The fundamentals of a refrigerant Heat Pump are dealt with in more detail than might be expected in a Study for a practical thermal energy recovery. The reason is that the data given by manufacturers are easy to misinterpret, with a possibility in specifying equipment which may be inadequate for the given conditions.

It appears that this application of a Heat Pump to the exhaust air (from a paper or pulp machine) after the conventional heat recovery is one of the first attempts to generate steam which substitutes steam from boilers and thus reduces operating cost.

There is a great potential for industrial low grade thermal energy recovery by the Heat Pumps. However, the manufacturer have to develop equipment suitable for the large industrial capacities.

July 1984 R.R. Kusec

1. SUMMARY

The intention is to recover and upgrade part of the thermal energy in the air vented from the News Machine and use it to produce low pressure steam. This steam is to be used for News Machine air heating and thus substitute the equivalent amount of steam generated in the electric boilers.

Different types of Heat Pumps (such as refrigerant compression, absorption, chemical, direct vapour compression, and thermocompression) were studied and it was found that, for the given temperature range, the optimum system is the refrigerant compression Heat Pump generating sub-atmospheric pressure steam, which is then compressed by centrifugal compressors to two pressure levels.

The choice of the system is based on the characteristics of the Heat Pumps working mediums. In this case, the refrigerant compression cycle uses DICHLORTETRAFLUOROETHANE (R114), and water (steam) for the centrifugal compressors. The refrigerant cycle raises the temperature from 43°C (110°F) to 93°C (200°F), and then the steam cycle lifts it in the first stage of compression to 91°C (195°F), and in the second stage to 129°C (263°F). The overall Coefficient of Performance (COP) = 2.9.

Savings on the consumed energy is 34,020 MWh and savings on the energy cost is $600,000 per year. Installation cost is about $2,200,000.

2. INTRODUCTION

The goal is to apply a suitable Heat Pump system to recover some of the waste heat carried by the exhaust air from the News Machine hood and produce low pressure steam. The low pressure steam is generated in a reboiler and then is compressed to substitute steam for the hood make-up air heater, and for the pocket ventilator air heater.

The "conventional" heat recovery from the machine hood exhaust is thermodynamically a good and useful method and therefore, only recovery of the residue thermal energy has been considered.

When analyzing potential waste heat recovery we must distinguish between the heat content and the possible recoverable heat energy.

The heat content of a stream is defined and measured from 0°C (32°F), (the corresponding specific heat or enthalpy is taken from the published tables for thermodynamic properties). When expressed in that manner, the amount of the heat energy may appear very large. However, in the majority of cases, we cannot use all of the heat because for most processes or for air heating, we are limited to temperature of 20°C minimum. For most processes the usable heat is at temperatures above 60°C. Therefore the possible recoverable heat energy is measured to the applicable temperature which can be used for a process.

For example, in our case the exhaust air after the conventional heat recovery is at 49°C (120°F) (saturated) with a heat content of:

11202 lb/min x 60 x 111.65 Btu/lb d.a. x 1.0551
= 79,177,023 kJ/h (or 22 MW)*

However, for practical purposes, only a small part of the 79 x 10 kJ/h can be used. If the air is cooled by a reasonable temperature difference of 5.6°C (10°F), the actual recoverable heat is:

11202 lb/min x 60 x (111.65 - 84.535) x 1.0551
= 19,228,706 kJ/h (or 5.3 MW),

which is only about 24% of the heat content.

This recoverable amount of waste heat is at a relatively low temperature and is not useful for our purpose. To implement this available heat, the temperature of the medium is raised to an acceptable level by a Heat Pump.

Different methods and mediums for upgrading the low level heat were analyzed. In general, for any waste heat recovery, the listed methods should be considered. The methods are listed in order of simplicity and cost of installation and operation, starting with the optimum one.

1. Direct heat exchangers.
2. Compression of low pressure waste steam.
3. Use of a low pressure steam generated by "dirty" steam, or hot liquid in a reboiler.
4. Compression of steam from a reboiler.
5. Freon cycle Heat Pump producing hot water.
6. Compression of low pressure steam (sub-atmospheric) from a Freon cycle Heat Pump reboiler.

* These figures are based on the subject Newprint Machine.

7. Heat Pump using methanol (or equivalent) producing low pressure steam which can be used directly or compressed to a higher pressure. (Due to the inherent danger and still as an experimental Heat Pump, this method has not been considered.)

The above methods were analyzed and as a result, the following was selected:

A two (or three) stage Freon cycle produces sub-atmospheric pressure steam, which is compressed to two levels of pressure. The lower pressure steam is substituting boiler steam for the Hood Make-up air heater, and steam from the second stage compression is used for Pocket Ventilation air heaters.

For the refrigerant cycle, a limit of 99°C (210°F) generating steam at 93°C (200°F), was selected because this temperature can be achieved by a reasonable design.

Refrigerant R114 (by E.I. du Pont de Nemours & Company, Inc.) is the only one currently available which can be used to achieve the above condition. The number of compression stages depends upon the characteristics of the centrifugal compressor available to suppliers. The sub-atmospheric pressure steam generated by the Freon Heat Pump is then compressed with centrifugal type compressors, which for the given condition, can develop a compression ratio of up to 2 : 1 in one stage.

3. REVIEW OF THE HEAT PUMP SYSTEMS

Here, we are reviewing only those types of Heat Pumps which are proven as a system, r their components are rendered operational. A Heat Pump is a device which upgrades low temperature heat source to a higher one which will deliver sufficiently high temperature for a process. This upgrading is achieved by input of mechanical energy which is converted to thermal energy. The mechanical energy (compression) is obtained most commonly from electrical energy (electric motor).

To do this, a fluid is used which evaporates at low temperatures. By evaporating the working fluid, heat is transferred from the source to the fluid. Next the vapour gas is compressed to achieve a higher temperature and then by condensing the vapour gas, heat energy is transferred to a process fluid at a higher temperature than the source. The amount of heat transferred at the higher temperature is equivalent to the heat picked up from the low temperature source plus electrical energy required for compression, less some mechanical, electrical, and thermal losses.

For low range temperatures (up to 200-240°F) different types of refrigerants are used, such as R12 and R114. For higher temperatures, water (steam) can be used. The following table compares some of these Heat Pump working fluids.

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COMPARISON OF WORKING MEDIUMS FOR HEAT PUMP APPLICATION**

	R12	R114	STEAM (WATER)	(R22)
110°F				
EVAPORATION ENTHALPY BTU/LB	54.31	51.85	1031.6	70.05
VAPOUR - ENTHALPY BTU/LB	87.84	86.16	1109.5	112.5
PRESSURE PSIA	151.11	53.78	1.27	241.04
SPECIFIC VOLUME CU FT/LB	0.267	0.596	265.4	0.222
160°F				
EVAPORATION ENTHALPY	44.37	45.91	1002.3	52.32
VAPOUR - ENTHALPY	91.0	93.21	1130.2	112.26
PRESSURE	279.82	109.63	4.74	444.53
SPECIFIC VOLUME	0.136	0.293	77.29	0.107
200°F				*
EVAPORATION ENTHALPY	32.08	40.15	977.9	21.99
VAPOUR ENTHALPY	91.29	98.34	1145.9	102.85
PRESSURE	430.09	178.42	11.53	686.4
SPECIFIC VOLUME	0.077	0.174	33.64	0.047
220°F	*			*
EVAPORATION ENTHALPY	21.79	36.75	965.2	
VAPOUR ENTHALPY	89.04	100.6	1153.4	
PRESSURE	524.43	222.84	17.19	
SPECIFIC VOLUME	0.053	0.019	23.15	
240°F	*			*
EVAPORATION ENTHALPY		32.79	952.2	
VAPOUR ENTHALPY		102.51	1160.5	
PRESSURE		275.25	24.97	
SPECIFIC VOLUME		0.104	16.32	
260°F	*			*
EVAPORATION ENTHALPY		27.27	938.7	
VAPOUR ENTHALPY		103.8	1167.3	
PRESSURE		337.09	35.43	
SPECIFIC VOLUME		0.078	11.76	

* Not recommended or not applicable

** Data in SI units are not available at this time

Refrigerant R12 is used for temperatures below 93°C (200°F), where R114 can be used for higher temperature. Note that R114 has approximately twice the specific volume as R12 for the same temperature condition, however R12 has approximately double the pressure than R114. Steam (water) is suitable for temperature above 82°-93°C (180°-200°F) and is limited only by available compressors (and of course, by the critical point of water). At low temperatures, the specific volume of steam is very large when comparing it to refrigerants, which inhibits the design and the application of such compressors.

The available Heat Pumps could use different methods of heat transfer from the source and to the process fluid. Some of them are:

a) Indirect, using water (or glycol solution) as intermediary heat transfer medium through heat exchangers between the refrigerant evaporator and condenser.

b) Indirect, using water as intermediary heat transfer medium which is in direct contact with source (gas, air).

c) Direct heat transfer from the source to the refrigerant evaporator and from the refrigerant condenser to a process fluid.

The following Figures No's 1, 2, and 3 show these three methods where the source is saturated air at 49°C (120°F), and steam is generated at 80 kPa(abs) (11.6 psia), 93°C (200°F).

Due to the desired capacity, the Heat Pump will use centrifugal type compressor. The characteristic of a centrifugal compressor is such that the compression ratio is less than 2 : 1 and, therefore more than one stage compression is needed to reach the desired temperature.

During the compression the vapour (gas) becomes superheated and is desuperheated before it is used in a process or before it goes through a second stage compression. Without the cooling, the sensible heat content of the vapour becomes a significant part of the total enthalpy.

Since the latent heat carries the major portion of the heat to be transferred, an additional heat transfer area must be provided to absorb the sensible heat of the superheated vapour before the latent heat can be utilized.

If the system shown in Figure No. 1 is used, a three stage compression, with interstage desuperheating, would be needed to achieve 104.4°C (220°F) minimum temperature of the refrigerant vapour in the condenser. The system would be complicated and would use more electrical power than a direct contact type Heat Pump. Refrigerant temperature must be raised for 72.2°C (130°F), from 32.2°C (90°F) to 104.4°C (220°F).

In Figure No. 2, warm water is sprayed into the air stream, collected and circulated through the evaporator. The refrigerant may need two or three stages of compression, and its temperature is raised by 66.8°C (120°F), from 32.2°C (90°F) to 99°C (210°F).

Figure No. 3 shows a system where the refrigerant evaporator is placed in the stream of hot air (source). Refrigerant temperature rise is only 61.2°C (110°F) however, still needs a two-stage compression to reach the desired final temperature of 99°C (210°F). A three stage

compression may be required in order to reduce the size of the evaporator. This scheme should use the least power and is the simplest of all others presented.

During the extraction of heat from the 49°C (120°F) saturated air, about 200 lb/min (24 gpm) of condensate will be released at a temperature of around 43°C (110°F).

For the refrigerant cycle a limit of around 93°C (200°F) is selected, because this temperature can be reached with a reasonable system design. In all cases the heat transfer takes place only when there is a temperature difference. Therefore when water is used as a heat transmitting medium it leaves the heat exchanger at a somewhat higher temperature then the heated fluid. In Case No. 1, the heating water enters the reboiler at 105°C, and leaves at 99°C. The condensate in the reboiler evaporates at 93°C. In Cases No. 2 and No. 3 the refrigerant enters as a gas and leaves the reboiler as a liquid at the same temperature of 99°C transferring its heat of evaporation (latent heat) to the boiler water (condensate).

A higher temperature steam can be achieved by direct compression of the 93°C steam (80 kPa(abs), 11.6 psia) generated by Freon cycle in the reboiler.

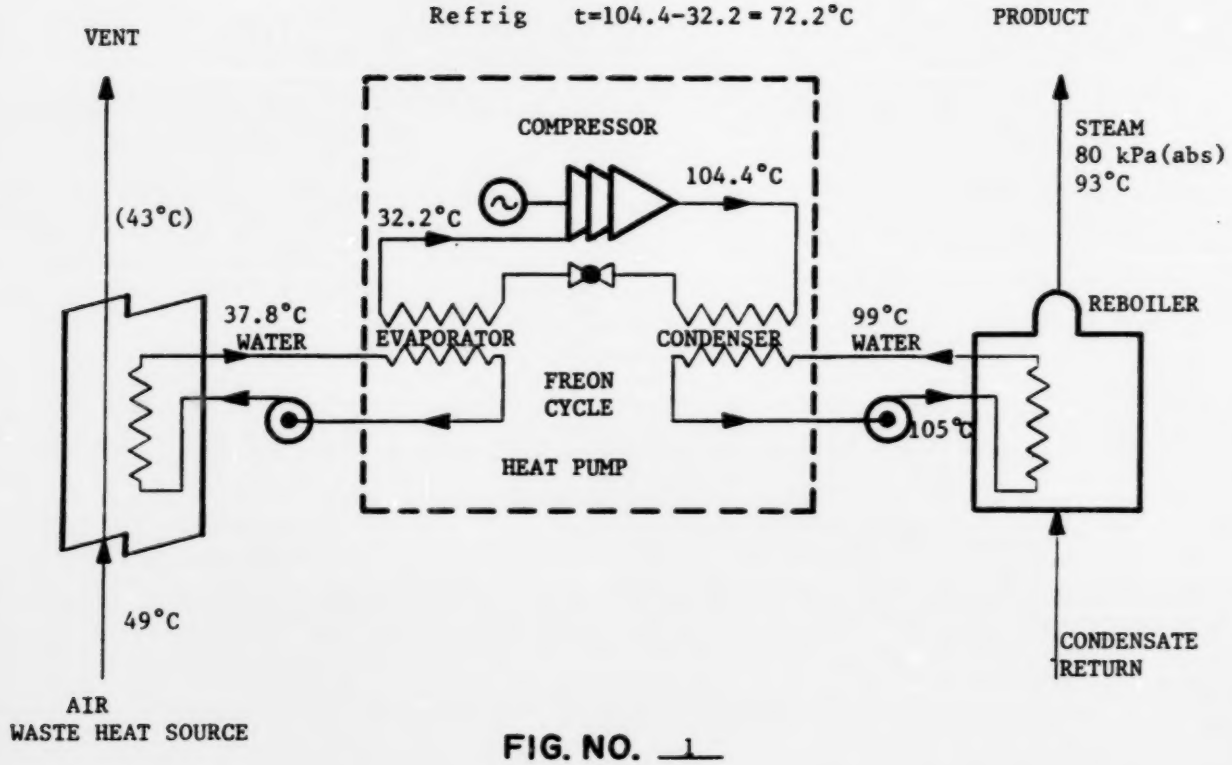


FIG. NO. 1

HEAT PUMP WITH INDIRECT HEAT TRANSFER

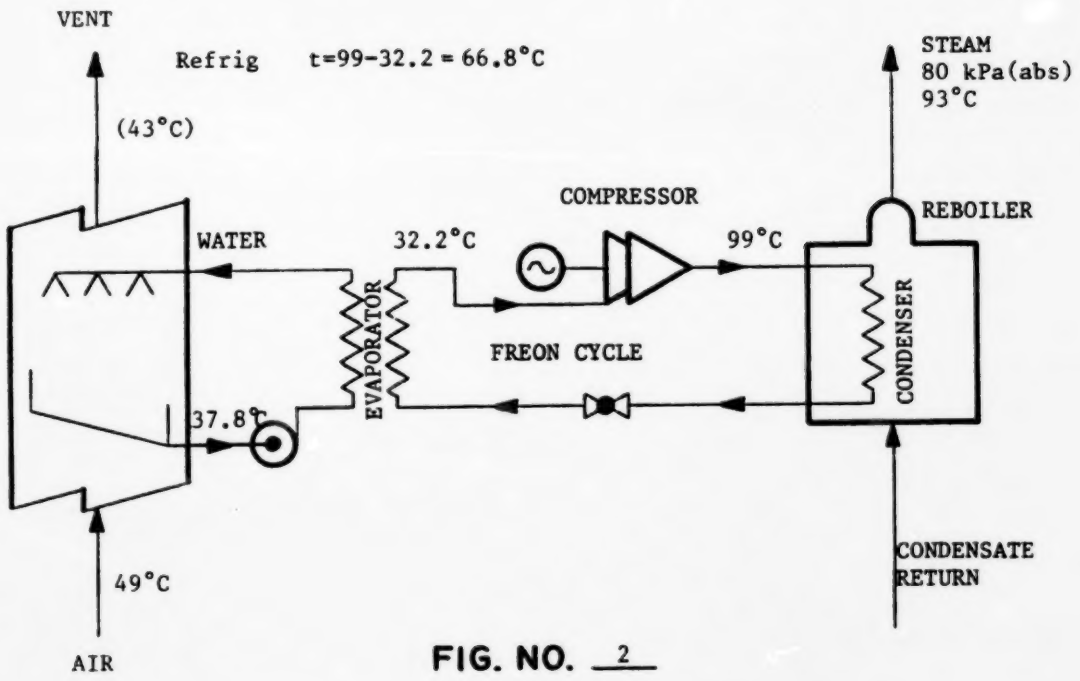


FIG. NO. 2

HEAT PUMP WITH INDIRECT/DIRECT HEAT TRANSFER

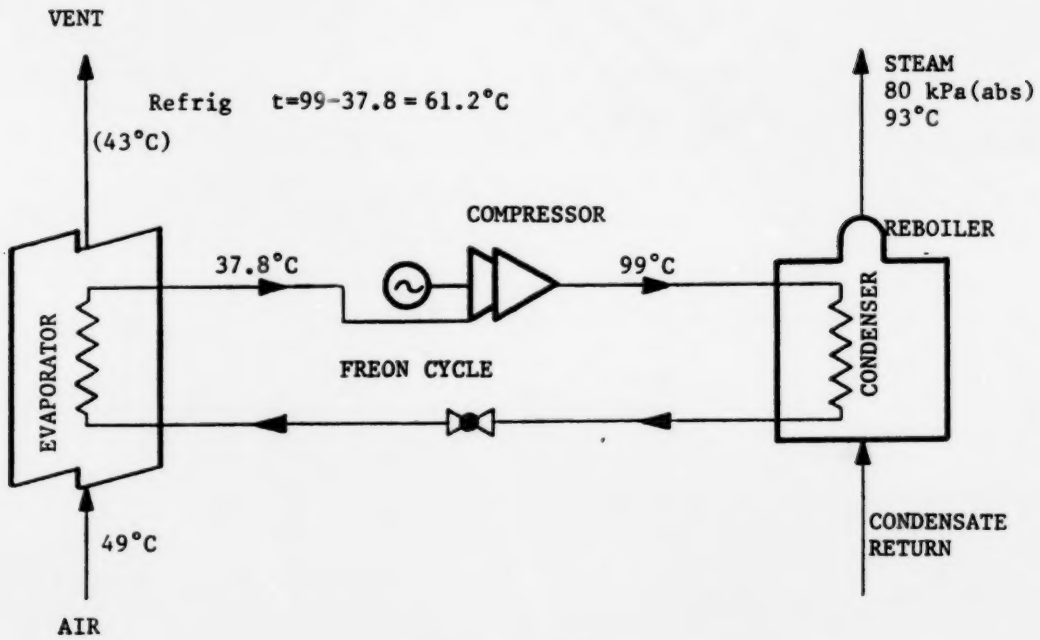


FIG. NO. 3

HEAT PUMP WITH DIRECT HEAT TRANSFER

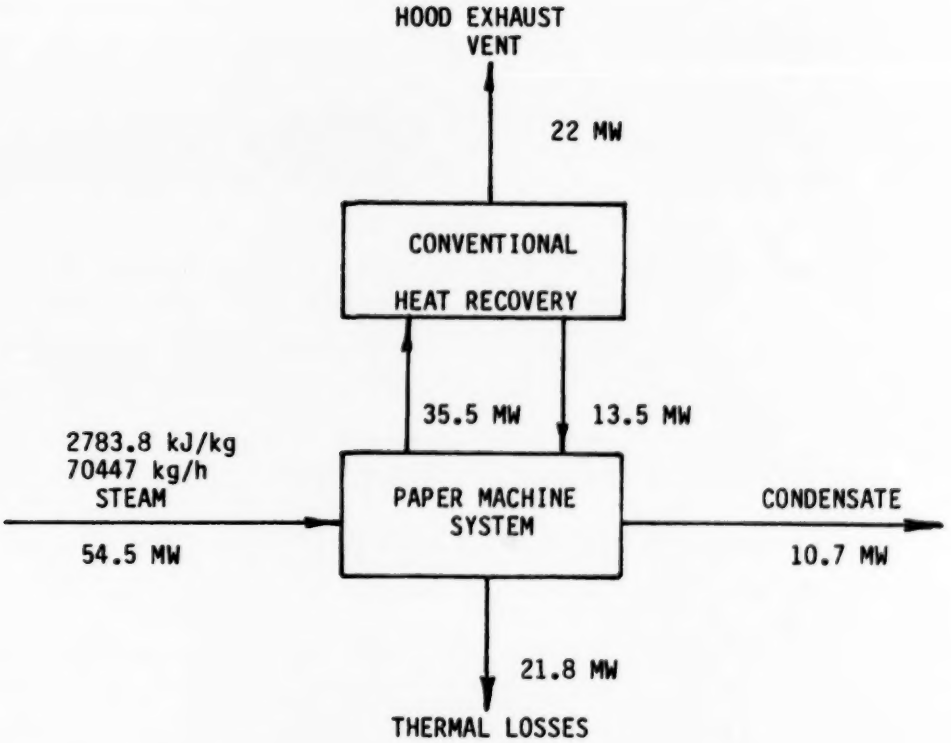
4. WASTE HEAT SOURCES ON THE NO. 3 NEWS MACHINE

There are many streams of fiber, water, and air leaving News Machine complex which consist of stock preparation and paper drying. Each of the streams contain a certain amount of thermal energy which is part of the overall heat and mass balance. The main heat energy input is by steam and the main output by exhaust air. The remaining heat is taken out in the form of warm water which is used in other processes, plus heat losses (e.g. cooling-off of paper leaving dryer, radiation, convection, cold air infiltration, etc.).

Figure No. 4 indicates the overall heat balance of the News Machine.

Heat energies leaving the machine, which could be practicably recovered are carried out by exhaust air and water. However, most different streams of warm water leaving the machine room are used somewhere else, and therefore, this heat is not available for recovery.

Air exhaust from the machine carries vapour from evaporation of water from drying the paper and consequently contains a large amount of heat energy, e.g. 35.5 MW. Out of this, 13.5 MW is normally recovered in a "conventional" heat recovery system.



ENERGY BALANCE (MW)

IN		OUT	
STEAM	54.5	VENT	22
		CONDENSATE	10.7
		TH.LOSSES	21.8
			54.5

FIGURE 4 - PAPER MACHINE OVERALL ENERGY BALANCE
WITH CONVENTIONAL HEAT RECOVERY

The hood exhaust air (5083 kg/min) at a condition of 78/58°C (173/137°F) is drawn through the economizers, where it preheats clean air from the mezzanine floor, then heats ethylene glycol water solution, and finally heats warm water from 27°C (80°F) to 40°C (104°F).

After all the heat recovery mentioned above, the hood exhaust air is discharged to atmosphere at 49°C (120°F) sat., and carries some 22 MW of thermal energy, of which a part could be recuperated. The amount of the recoverable thermal energy depends upon the practically achievable temperature drop of the source. With a reasonable temperature reduction of, for example, 5.6°C (10°F), the potential available energy is 5.34 MW. However it must be upgraded to a higher temperature to make it usable. The upgrading is done by input of a certain amount of the external energy in the form of vapour compression. This external energy is added to the recoverable amount of energy and thus, increases the total usable energy.

5. UTILIZATION OF THE RECOVERED THERMAL ENERGY

The Pocket Ventilation units use 5000 kg/h steam and the Hood Make-up unit uses 2936 kg/h of steam at pressure of 517 kPag (75 psig). The net heat transferred to heat the air is 16.54×10^6 kJ/h or 4.6 MW.

The Heat Pump system produces low pressure steam which substitutes the 517 kPag steam, but the amount of energy remains the same, that is, 4.6 MW. This energy is carried by 4,787 kg/h of 262 kPa(abs) and 2739 kg/h of 143 kPa(abs) pressure steam. Since the Heat Pump system uses 1.6 MW, only 3 MW is taken out of the hood exhaust, where 5.34 MW is available as discussed in Section 4. The remaining 2.34 MW is lost with the vented air.

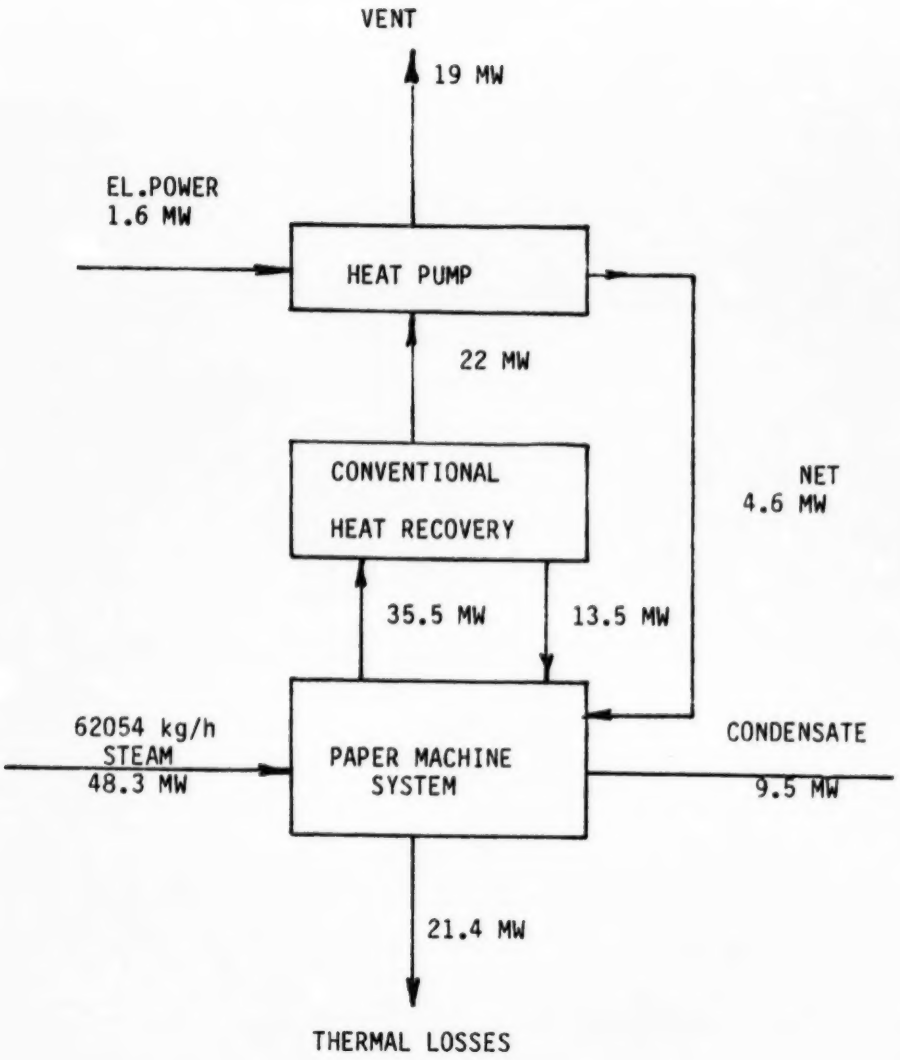
Figure No. 5 illustrates the overall energy balance of the News Machine complex when the Heat Pump is used.

For economic heat recovery, the most promising application is for continuous heat energy users.

On a News Machine, there are three major continuous users of steam:

a. **DRYER CYLINDERS**
b. **POCKET VENTILATION AIR HEATERS**
and c. **HOOD AIR MAKE-UP HEATER**

As it will be shown in the proceeding description of the process, only Pocket Ventilation and Hood Air Make-up can be considered for application of low pressure steam produced by a Heat Pump system using hot waste air as a source of low-grade heat.



ENERGY BALANCES (MW)

IN		OUT	
STEAM	48.3	VENT	19
EL. POWER	1.6	CONDENSATE	9.5
		TH. LOSSES	21.4
TOTAL	49.9	TOTAL	49.9

FIGURE 5 - PAPER MACHINE OVERALL ENERGY BALANCE WITH HEAT PUMP APPLICATION

a. **Dryer Cylinders**

The temperature range of the steam used in different sections of the paper drying machine is from 130°C (267°F) to 150°C (303°F). The main steam supply is at 517 kPag (75 psig), which is used directly in two cylinder sections at 379 kPag, 150°C (55 psig, 303°F). The other four sections are receiving flash steam from condensate of the preceding sections. The flash steam is boosted to the desired pressure by thermocompressors using 1069 kPag (155 psig) steam. Therefore, the steam and condensate system of the dryer is interdependent and any steam which may be brought in will have to be at a minimum pressure of 379 kPag (55 psig). Any other low pressure steam which would be introduced into low pressure sections could upset balance of the system (as designed for this machine). Upgrading the available waste heat from 49°C (120°F) to 150°C (303°F) would require more than three stages of compression for the steam cycle of the Heat Pump system. Technically this can be done but it would be prohibitive when the economics of such an installation is considered.

b. **Dryer Cylinders Pocket Ventilation**

Air for pocket ventilation is heated to 121°C (250°F). To do this, there are two groups of air heaters. The first one uses flash steam from the condensate tank at 67 kPag (9.7 psig) and the second one uses 517 kPag (75 psig) boiler steam.

Flash steam is expected to be up to 1131 kg/h (2492 lb/h) for each unit (there are two P.V. units). However, it is quite possible that flash steam will not be available. In this case, steam at 517 kPag (75 psig) will supplement the heat from flash steam.

This air heater will use up to 2503 kg/h (5519 lb/h) of 517 kPag (75 psig) steam per each P.V. unit or total of 5007 kg/h when flash steam is not available. (See Figure No. 6). The quoted steam flows are calculated taking air from the economizer at 49°C (120°F) saturated.

The Heat Pump system supplies steam (for P.V.) at 262 kPa(abs), 23 psig, which replaces steam at 517 kPag, 75 psig. The quantity of low pressure steam is reduced because the latent heat is larger than for the high pressure steam. The low pressure steam flow is 4787 kg/h instead of 5007 kg/h for the 517 kPag (75 psig) steam.

c. **Hood Make-up Air Heating**

Room air (from the mezzanine floor) is preheated from 32°C (90°F) to 47°C (116°F) by the exhaust from the machine. After this, the air temperature is raised to 91°C (195°F) using the boiler steam at 517 kPag (75 psig) pressure. For the air flow of 2200 kg/min (4858 lb/min) it requires 2936 kg/h (6459 lb/h) of 517 kPag steam.

Since the final temperature of the air is 91°C, steam at 143 kPa(abs), 6 psig is used. Here again, the heat transfer area is increased by additional steam coils, and low pressure steam flow is 2739 kg/h instead of 2936 kg/h of high pressure steam.

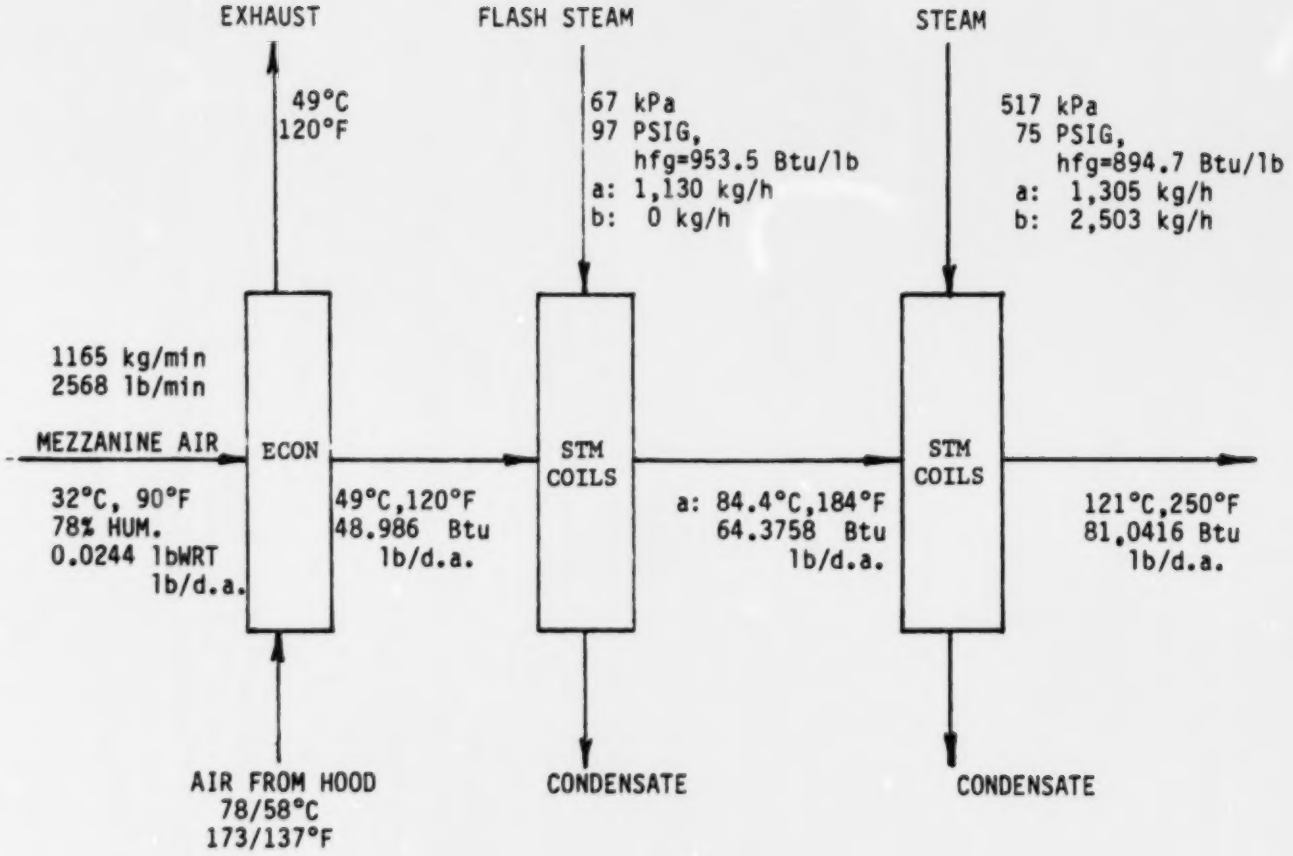


FIGURE NO. 6 - POCKET VENTILATION UNIT
a: Flash Steam Used
b: Flash Steam Not Available

6. PROPOSED HEAT PUMP SYSTEM

The proposed Heat Pump system consists of a refrigerant cycle and steam compression cycle. The product of the system is low pressure steams at 143 kPa(abs), 110°C (230°F), and at 262 kPa(abs), 129°C (264°F).

To compensate for the lower temperature of the steam from the compressors, the heat transfer area is increased. This is achieved by adding coils in series with those that are already supplied.

The steam pressures delivered by the centrifugal compressors were selected such that the temperature difference between steam and air will give a reasonably sized heat exchanger, which could be fitted into already designed equipment, and at the same time, will comply with the compression ratios which can be achieved by the centrifugal compressors. As a consequence, the pressure of the reboiler was determined at 80 kPa(abs), 93.5°C (200°F) followed by two stages of compression.

In order to generate steam at 93.5°C, the refrigerant in the condenser is at 98.5°C (209.4°F). The waste heat source (exhaust) is at 49°C (120°F), and the refrigerant evaporates at 36°C (97°F). To achieve 98.5°C in the Freon condenser, a three stage centrifugal compressor is needed. This particular temperature for the evaporator is selected by a specific Heat Pump manufacturer.

Some other manufacturer may choose a higher temperature and then reach 98.5°C by two stages of refrigerant compression. However, the size of the heat exchanger (evaporator) will be much larger than the previous one. Even with the

temperature difference of 13°C (= 49-36) the size of the evaporator is about 7.4m x 5m (24 ft x 16 ft) for the capacity considered in this Study.

Figure No. 7 is a schematic showing the entire plan for the exhaust waste heat recovery and its usage.

The energy balance is shown on Figure No. 5. the overall coefficient of performance (COP), defined as:

$$COP = \frac{\text{USEFUL THERMAL ENERGY}}{\text{ENERGY INPUT BY COMPRESSORS}},$$

$$\text{is } COP = \frac{4.6 \text{ MW}}{1.6 \text{ MW}} = 2.9$$

The machine hood has four vents which are equally spread along the full length of the paper dryer. Vent stacks are connected to a collecting header which is attached to the housing of the refrigerant evaporator. Also, each vent stack is provided with a damper to direct the exhaust air into the header. Here only hot air from three stacks is collected in a common header, and the fourth one, as an option, is used to preheat boiler feed water.

Figure No. 7 presents the entire scheme for the exhaust waste heat recovery and its usage.

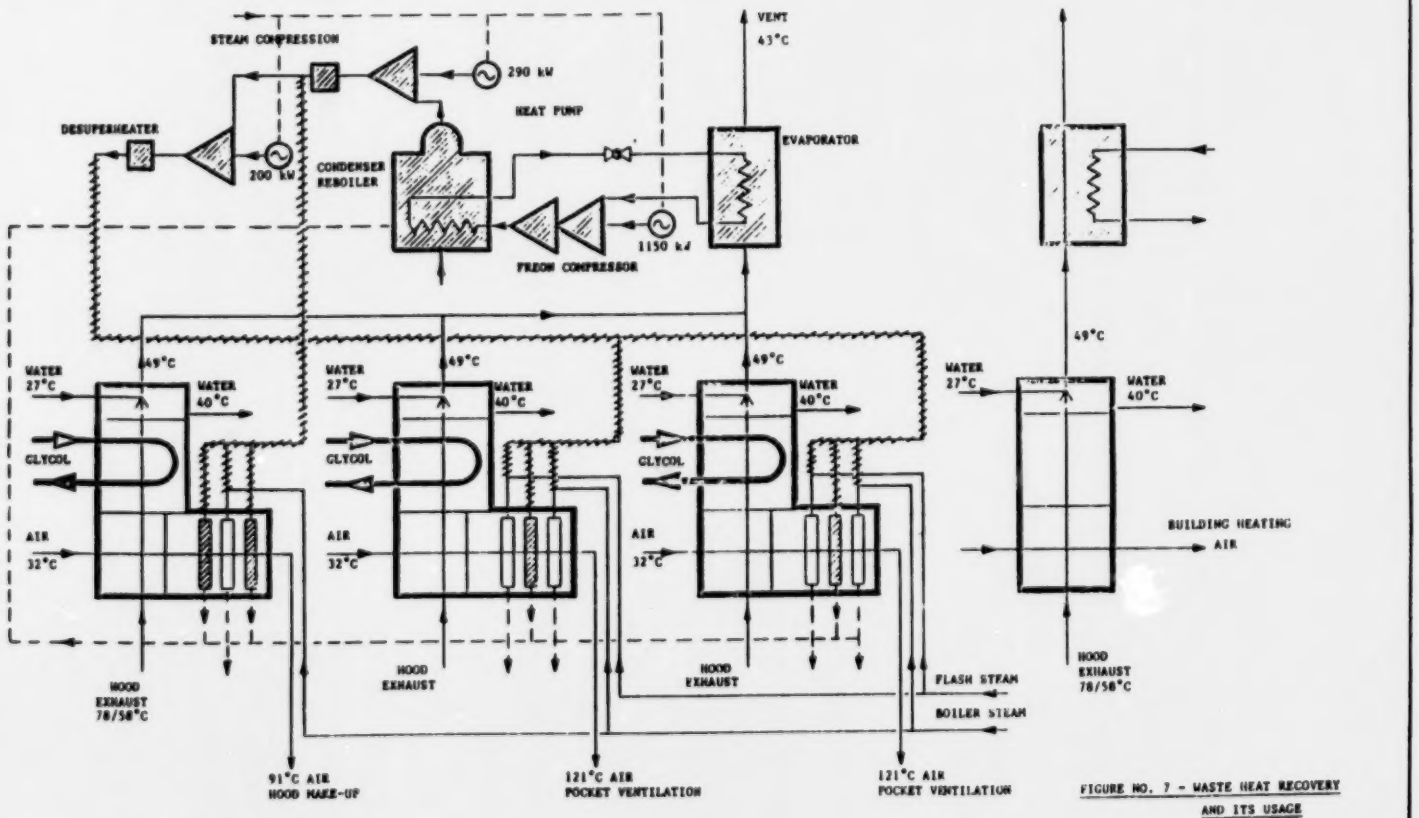


FIGURE NO. 7 - WASTE HEAT RECOVERY AND ITS USAGE

7. VENDORS' PROPOSED SYSTEMS

An inquiry was issued for supply of a complete system to upgrade waste heat carried by the exhaust air as low pressure steam.

Proposals were received as follows:

a) two (2) for a complete system
b) one (1) for refrigerant cycle only
c) one (1) for steam compression only
d) one (1) rejection

a.1. This Vendor's design uses two stage flashing of the refrigerant with bleed-off from the tanks to the compressor. Freon R114 is continuously recirculating through the evaporator air coils from a tank, using a pump. This feature ensures maximum heat recovery from the source (exhaust air). The Freon vapour going to the compressor is taken from the recirculating tank which acts as a droplet separator as well.

The condenser-reboiler is in the form of a horizontal heat exchanger. Refrigerant vapour is on the shell side and the evaporating water is on the tube side. The wet steam is discharged into a vessel where water droplets are separated from steam, which goes into the first stage of the steam compression cycle.

The operating power is

Freon Cycle	1150 kW
Steam Cycle	614 kW
TOTAL	1764 kW (expect 1.6 MW)

Note: It appears that these steam compressors have low efficiency. A more efficient compressor can be obtained from another supplier.

The steam compression cycle consists of two separate compressors each with its own motor drive. Between the stages the steam is desuperheated. (See Figure No. 3). The overall COP = 4.6/1.6 = 2.9.

a.2. In this proposal, the refrigerant is flashed in one stage and then the Freon R114 is cooled in a heat exchanger by the refrigerant vapour drawn by the compressor. The evaporator is a once through type where the refrigerant is in the shell and the exhaust air flows through the vertical tubes.

The condenser-reboiler is a heat exchanger where the refrigerant vapour is condensing in the tubes, and the water is evaporating on the shell side. (this is the opposite to a.1. proposal).

The steam cycle consists of two compressors driven by a common motor drive.

The operating power is for

Freon Cycle	1100 kW
Steam Cycle	486 kW
TOTAL	1586 kW

The overall COP = 4.6/1.6 = 2.9.

b. This Vendor quoted a Freon cycle only because he does not have suitable equipment for the steam cycle.

The Freon R114 cycle consists of a single first stage Heat Pump followed by a two stage Heat Pump with two separate compressors, or a total of three compressors.

Heat from the air stream is absorbed by a water heat exchanger, and the system delivers water at 102°C (215°F) and returns at 90°C (195°F). The process is similar to the one shown in Figure No. 1.

To the above Heat Pump a steam compressor must be added by another supplier.

The operating power is

	Freon Cycle	590 kW
		546 kW
		514 kW
	Steam Cycle	501 kW
	TOTAL	2151 kW

The overall COP = 4.6/2.2 = 2.1.

c. This is an offer to supply a steam compressor unit consisting of two centrifugal compressors driven by one motor through a common gear. Operating power is 501 kW.

8. EFFECT OF HEAT PUMPS ON ENERGY BALANCE AND ON THE ECONOMICS OF ITS INSTALLATION

The overall effect of a Heat Pump on the energy balance is reduction of fuel to the boilers and an increase of energy input for driving the compressors of the Heat Pump system. The reduction of energy input to the boiler is greater than the additional energy needed for driving the Heat Pump compressors.

In this Study, the fuel for the boilers is electricity and therefore, the economics of the Heat Pump application depend on the reduction of the operating cost by using less electrical energy, and on the cost of the Heat Pump installation.

Steam, condensate, and boiler feed flows are shown on the flow diagrams Figure No. 8 and Figure No. 9. The quoted flows are for the rate of production of 621 OD.MT/day at 100% efficiency of the News Machine. The data may vary somewhat from the actual but any difference is not significant for our purpose where we compare the electrical demand with or without the Heat Pump employment.

Figure No. 8 is the flow diagram without the Heat Pump application for average winter conditions.

Exhaust air from the Paper Machine flows through a heat recovery system. About 38% of its heat content is used for air preheating, glycol water solution heating, and hot water make-up for the process. After that, it is vented to the atmosphere at 49°C (120°F) in a saturated condition and still carrying a substantial amount of thermal energy.

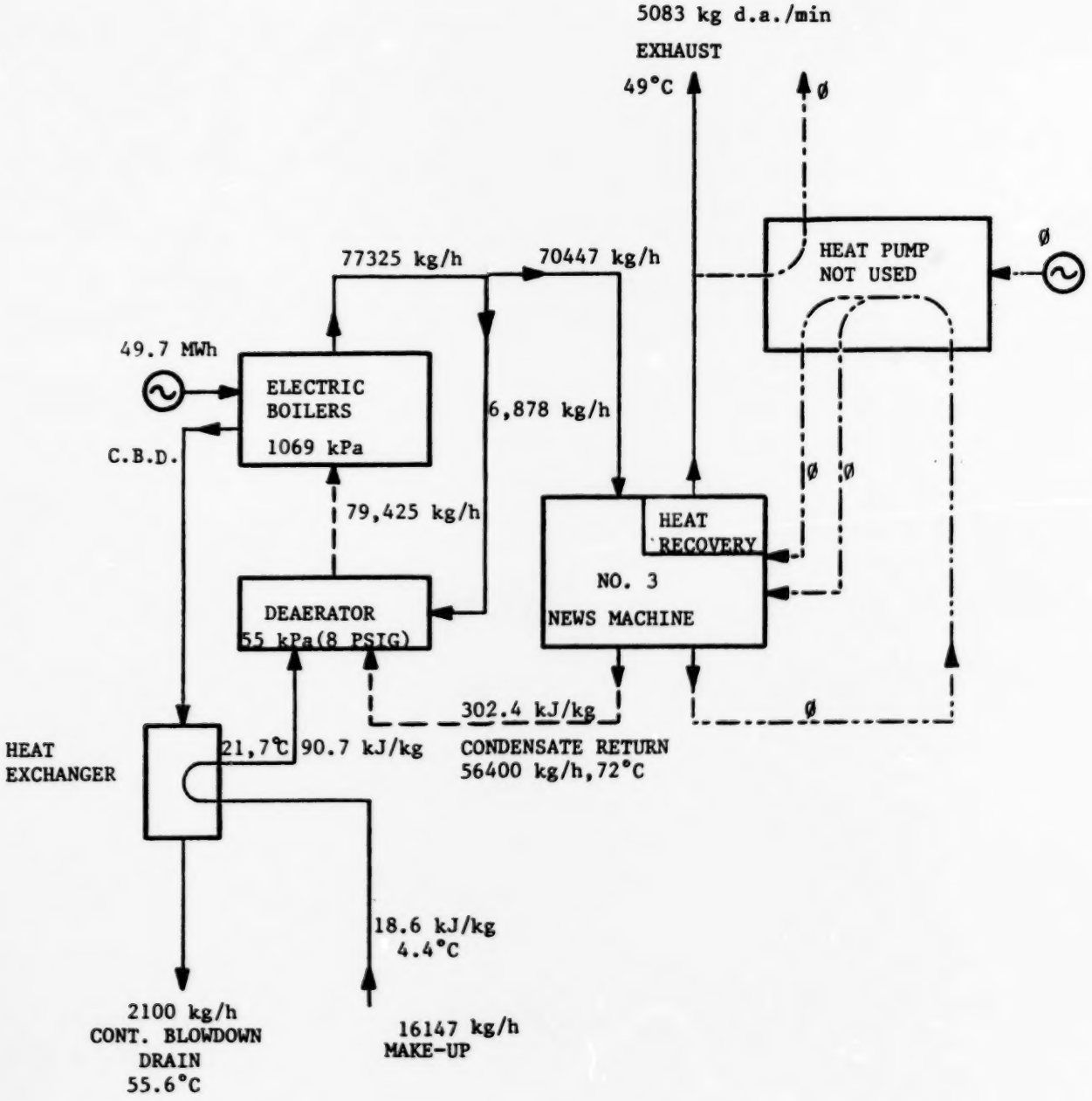


FIGURE NO. 8 - STEAM GENERATION FLOW DIAGRAM
WITHOUT HEAT PUMP APPLICATION

ENERGY AND MASS FLOW BALANCE

STREAM	FLOW kg/h	FLOW lb/h	ENTHALPY kJ/kg	ENTHALPY Btu/lb	HEAT 10^6 kJ/h	HEAT 10^6 Btu/h
OUT						
STEAM TO P.M., 1069 kPag	70,447	155,366	2,783.7	1195.6	196.10	185.64
CONT. BLOW-DOWN	2,100	4,631	232.4	100.0	0.49	0.46
TOTAL	72,547				196.59	
IN						
CONDENSATE RETURN	56,400	124,362	302.3	130	17.05	16.17
MAKE-UP WATER	16,147	35,604	18.7	8	0.30	0.28
TOTAL	72,547				17.35	

ELECTRICAL ENERGY INPUT: $(196.59 - 17.35)10^6 = 179.24 \times 10^6$ kJ/h

or 49.7 MW

TABLE NO. 1 STEAM GENERATION FOR NO. 3 NEWS MACHINE

WITHOUT THE HEAT PUMP APPLICATION

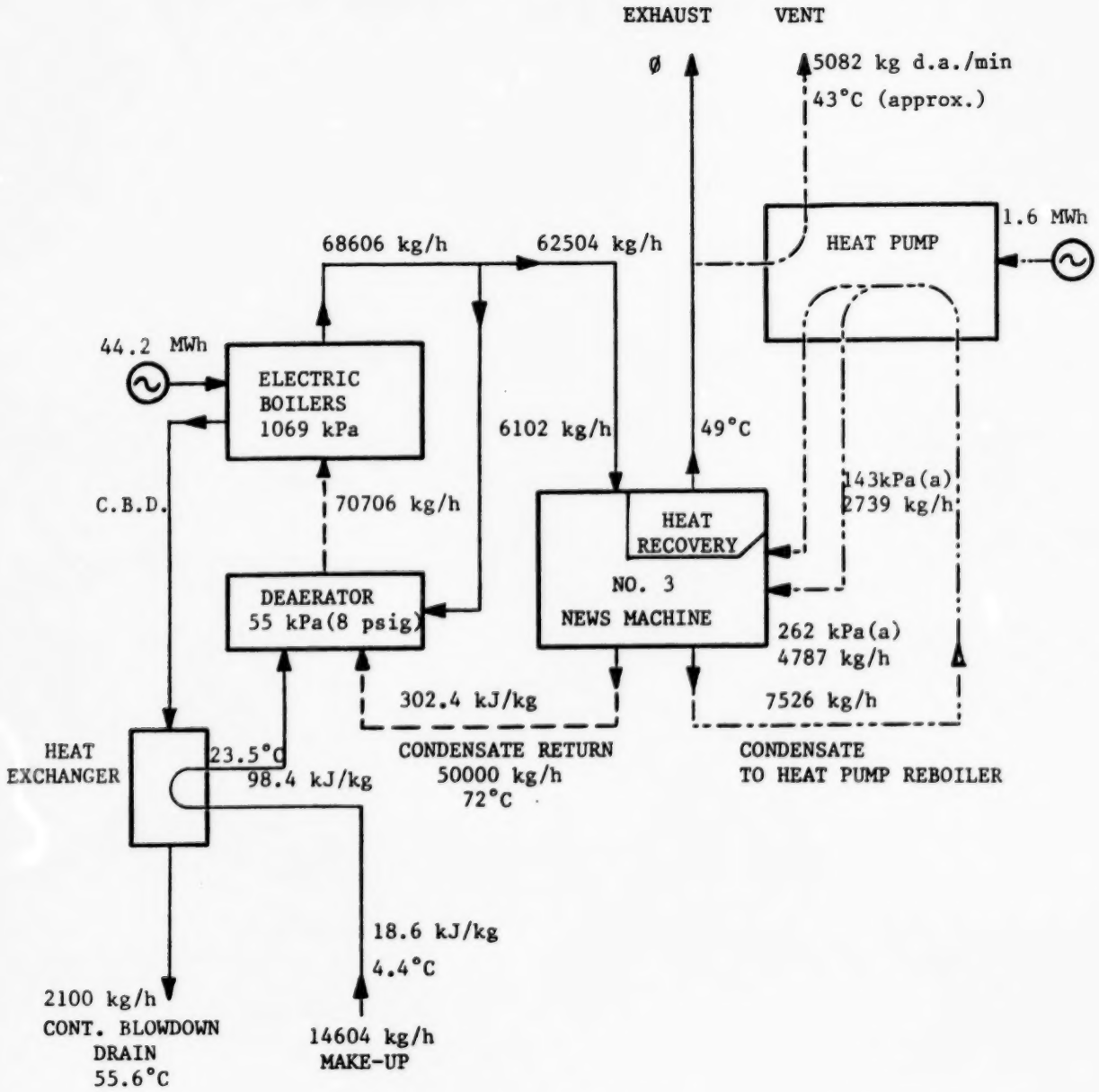


FIGURE NO. 9 - STEAM GENERATION FLOW DIAGRAM
WITH HEAT PUMP APPLICATION

SIMONS

ENERGY AND MASS FLOW BALANCE

STREAM	FLOW kg/h	FLOW lb/h	ENTHALPY kJ/kg	ENTHALPY Btu/lb	HEAT 10^6 kJ/h	HEAT 10^6 Btu/h
OUT						
STEAM TO P.M.	62,504	137,821	2783.7	1195.6	173.99	164.78
CONT. BLOW-DOWN	2,100	4,630	232.4	100.0	0.49	0.46
TOTAL	64,604				174.48	
IN						
COND. RETURN	50,000	110,250	302.3	130	15.12	14.33
MAKE-UP WATER	14,604	32,202	18.7	8	0.27	0.26
TOTAL	64,604				15.39	

ELECTRICAL ENERGY INPUT:

$$(174.48 - 15.39) \times 10^6 = 159.1 \times 10^6 \text{ kJ/h (44.2 MW)}$$

BOILERS	44.2 MW
HEAT PUMP	1.6 MW
TOTAL	45.8 MW

TABLE NO. 2 - STEAM GENERATION FOR PAPER MACHINE

WITH HEAT PUMP

Steam flow to the News Machine is 70,447 kg/h, condensate return at 80% is 56,400 kg/h, and the boiler feed make-up amounts to 16,147 kg/h. The deaerator operates at 55 kPag (8 psig) and uses 6878 kg/h of boiler steam. The boiler's continuous blowdown is taken as 2100 kg/h, which is around 3% of the generated steam.

The make-up water is preheated by the continuous blow-down drain from 4.4°C to 21.7°C, before entering the deaerator. The electric boilers have a power demand of 49.7 MW (178.9 x 10 kJ/h).

Figure No. 9 shows the effect of the Heat Pump system application. The source of the low grade heat is the exhaust air which still contains about 62% of thermal energy carried by the air before the conventional heat recovery. However, for our purposes, about 14% is recoverable and this only after it is upgraded to a higher temperature by a Heat Pump system producing low pressure steam. The low pressure steam is substituting boiler steam for the dryer cylinder pocket ventilation and for heating the make-up air for the machine hood. Exhaust air is vented to the atmosphere at about 43°C. By using Heat Pumps about 4.6 MW (net) is supplied to the machine in the form of low pressure steam. The net 4.6 MW consists of 3 MW heat recuperation from the exhaust and 1.6 MW electric power input through the Heat Pump compressors.

As a result, the electric boilers need to generate less steam and therefore the electrical load is reduced to 44.2 MW (instead of 49.7 MW without Heat Pump). The Heat Pump system uses about 1.6 MW, therefore the net reduction of power is 3.9 MW. (See Figure No. 8 and Figure No. 9).

All steam for the Paper Machine is generated by the electric boilers. The electrical energy rate for this purpose is

$0.016 per kWh. For Heat Pump drives, the rate is taken as $0.012 per kWh.

As shown on the heat balance diagrams, Figure No. 8 and Figure No. 9 and on Table No. 1 and Table No. 2, the energy demand without the Heat Pump is 49.7 MW and with the Heat Pump is 46.8 MW for the average winter condition.

During the summer, the Paper Machine itself will use the same amount of steam. The glycol-water solution is not required and therefore the exhaust air going through the Freon cycle evaporator is at a higher temperature which reduces power demand for the compressor by about 10 percent. Air for hood make-up and pocket ventilation from the mezzanine floor is now at 35°C (95°F) instead of 32°C (90°F). This will reduce the steam demand for hood make-up by 19 percent; and for pocket ventilation, 17 percent (approximately). Assuming that condensate flash steam is not available, which is very likely, the steam flow for hood make-up air heating is 2739 kg/h x (1-0.19) = 2219 kg/h or reduction of 520 kg/h. The required steam flow for the pocket ventilation air heating is 4787 kg/h in an average winter. In summer, the required steam flow is 17 percent less, or 4787 x (1-0.17) = 3973 kg/h for two units or a reduction of 872 kg/h. This represents a reduction of the electrical demand for the boilers of approximately 0.9 MW. Power for the Heat Pump is reduced, say, by 6 percent or 100 kW and the load is 1.5 MW (approximately), instead of 1.6 MW as in the winter.

During the summer conditions and <u>without</u> the Heat Pump, the boiler load is 49.7 - 0.9 = 48.8 MW.

At the same conditions but <u>with</u> the Heat Pump, the boiler load is 44.2 - 0.9 = 43.3 MW, and power for the Heat Pump compressor is 1.4 MW.

Summer - Energy usage **without** Heat Pump

48.8 x 4200 h/y = 204,960 MWh

and the cost is

204,960,000 x 0.016 = $3,279,360

Energy usage with Heat Pump

(43.3 + 1.5) 4200 = 188,160 MWh

and the cost is

[(43,300 x 0.016)) + (1400 x 0.012)] 4200

= $2,980,320

Winter - Energy usage **without** Heat Pump

49.7 x 4200 = 208,740 MWh

and the cost is

208,740,000 x 0.016 = $3,339,840

Energy usage with Heat Pump

(44.2 + 1.6) 4200 = 192,360 MWh

and the cost is

([44,200 x 0.016] + [1600 x 0.012]) 4200

= $3,050,880

Total for One Year

Without Heat Pump: Energy 413,700 MWh, Cost $6,619,200

With Heat Pump: Energy 380,5200 MWh, Cost $6,031,200

The saving on the energy cost amounts to about $600,000/year.

The Capital Cost has been estimated for this specific case as $2,200,000

9. CONCLUSIONS AND RECOMMENDATIONS

Of all the various methods to upgrade waste heat energy from Paper Machine hood exhaust for use in air heaters for the machine hood Air Make-up and the Pocket Ventilation, we found that the most suitable system is a combination of a Freon cycle followed by a steam compression cycle.

The following is a description of the recommended system:

A refrigerant cycle is used in two (or three) stages to generate steam in the reboiler at 80 kPa(abs), 11.6 psia.

Steam from the reboiler is compressed in the first stage compressor to 143 kPa(abs), then is desuperheated by spraying condensate, which increases the steam flow. From this, a part is condensed in the Hood Make-up air heater. The remainder goes through the second stage compressor raising the pressure to 262 kPa(abs), 38 psia. After the compressor, the steam is desuperheated using condensate, and is then used in the Pocket Ventilation air heater. Condensate from both users is returned to the reboiler.

Of the quotations received, the preferred system is the one where the refrigerant from the condenser-reboiler is flashed in two stages in large tanks, and the low temperature refrigerant for the evaporator is recirculated through a large tank from which dry saturated gas refrigerant is supplied to the compressor. The overall COP = 2.9.

APPENDIX

REFERENCES

a) MIDLAND-ROSS HOOD EXHAUST HEAT RECOVERY FLOW SCHEMATIC
b) MIDLAND-ROSS P.V. ROLL SUPPLY SYSTEM DWG. NO. 83-177-238,-239
c) MIDLAND-ROSS HOOD SUPPLY SYSTEM DWG. NO. 83-177-228
d) MIDLAND-ROSS P.V. SUPPLY UNIT SKETCH NO. S-177-248,-249
e) MIDLAND-ROSS HOOD SUPPLY UNIT SKETCH NO. S-177-236
f) H.A. SIMONS (INT'L) P & C DIAGRAM DWG. NO. AO-7079-524-028

NOTE: References are not included.